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ROYAL AIRCRAFT ESTABLISHMENT

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TECHNICAL NOTE No: R.P.D. 22

**A METHOD OF CALCULATING
THE WEIGHT AND DIMENSIONS
OF A TURBO PUMP FOR
ROCKET PROPELLANTS**

by

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ROCKET PROPULSION DEPARTMENT,

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4 A Method of Calculating the Weight and Dimensions of a
Turbo Pump for Rocket Propellants

by

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L5

SUMMARY

This note indicates methods by which the weight and size of a rocket motor turbo pump may be approximately obtained, in order to help the rocket motor designer to consider what type of fuel expulsion he should employ. Some examples are given, together with estimates of steam consumption. A suggestion is also given for reducing the turbo-pump weight.

1. Pumps.

I Saunders, D. J.

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1 Introduction

In order to design a particular rocket motor, the designer requires data on the nature of the propellants and their specific impulse, the thrust and the duration of the motor and whether repeatability of operation is required or not.

Though it is possible to design a combustion chamber, injector, etc., from this information it is not always obvious what type of propellant supply to the combustion chamber should be used to give low weight and simplicity or ease to manufacture.

The chief methods of removing the propellants from the tanks and supplying them to the combustion chamber under pressure, are by means of

- (1) A turbo-pump
- (2) Gas pressure in the tanks.

Some of the various ways of producing gas and applying it to the propellants have already been discussed¹ and this note confines itself to the first method with the object of enabling designers to assess the approximate weight, dimensions and general performance of a turbo-pump to meet any particular requirement.

Turbo-pumps were used in the German V.2 and the Walter 109-509 rocket motors, but these pumps were made on more or less conventional lines, and entailed rather complicated construction. The type of pump considered now is the Barske open impeller pump (with fluid seals) coupled with a bucket wheel type turbine². Fig.1 shows an example of this type of turbo-pump designed for an 1800 lb thrust rocket motor with a duration of two minutes. It is evident that the type of construction shown here greatly facilitates manufacture.

The most convenient layout of turbo-pump consists of a central turbine driving an oxidant pump on one side with a fuel pump on the other and having a common driving shaft to all three units. The pump shown in Fig.1 which is typical except for the shaft protruding from the fuel pump, will form the basis of the discussion. Fig.2 shows a single higher flow rate pump, the German 'Enzian' oxidant pump, whose characteristics will also be used in the discussion.

2 Estimation of Weight

The turbo-pump as a whole can be broken down into the following components as indicated in Fig.3.

- (1) Oxidant pump casing
- (2) Fuel pump casing
- (3) Oxidant pump impeller
- (4) Fuel pump impeller
- (5) Turbine wheel and shaft
- (6) Turbine and bearing casing including bearings
- (7) Fluid seals
- (8) Studs, bolts and other extras.

The method of calculating the weights of the pump casing will, of course, be the same for both fuel and oxidant, and it is only necessary

to substitute the relevant conditions. Similarly the impellers follow the same lines, but the turbine components will be quite different. The data which must be given (for both fuel and oxidant) are:-

- (1) Propellant throughput - Q lb/sec.
- (2) Propellant delivery pressure - p lb/sq.ft.
- (3) Propellant density - W lb/cu.ft.

Where necessary the symbols used will have suffix O or F to distinguish oxidant and fuel respectively. These and all other symbols are collected in Appendix I.

From the foregoing data and with certain assumptions, which are detailed later and can be justified by practical experience, it is possible to proceed. It is first necessary to estimate the pump rotation speeds and thence deduce the impeller and turbine radii. The weights of both impellers and casings are a function of these radii, whereas the turbine weight is dependent on an inverse function of rotational speed.

2.1 Calculation of Rotational Speed

The criteria for pump rotational speed are the axial velocity v of the fluid at the inlet of the pump with the largest throughput (usually the oxidant pump) and the peripheral velocity u at the inner radius of this pump's impeller.

Normally the value of v should not exceed 10 ft/sec, but higher values may be realized if the tanks are slightly pressurized or a screw type booster (as on the German 109-509) is used. In the latter case for weight calculation purposes v may be taken as a maximum of 15 ft/sec.

Consider the simple diagram of an open impeller pump shown in Fig.4. At the inlet

$$\pi r^2 v = \frac{Q}{W}$$

i.e.

$$r = 0.56 \sqrt{\frac{Q}{vW}} \text{ ft} \quad (1)$$

where r is the pump inlet radius in feet.

The peripheral velocity at the impeller inner radius

$$u = 2\pi r n \text{ ft/sec} \quad (2)$$

where n is the rotational speed in revolutions per second. At the present stage of development, the maximum value of u is considered as 170 ft/sec. Substituting this value in (2) and eliminating r by using (1)

$$n = 48.2 \sqrt{\frac{vW}{Q}} \quad (3)$$

or

$$n^2 = 2320 v \frac{W}{Q}$$

Fig. 5 and 6 show $\frac{W}{Q}$ plotted against $10^{-3} n^2$ for various values of v .

2.2 Weight of pump casing

Given the propellant throughput, pressure and density and the values of the rotational speed obtained from Fig. 5 or 6, the radius of each impeller can be obtained as follows:-

The delivery pressure p = static pressure + dynamic pressure
lb/sq ft

The static pressure = $\frac{W (\bar{U}^2 - u^2)}{2g}$ lb/sq ft

where \bar{U} is the peripheral velocity at the outer radius of the impeller in ft/sec.

This pressure is normally obtained fully.

The dynamic pressure is only partly obtained. ψ is a factor indicating the pressure recovery; it depends on the fluid viscosity and the internal surface finish of the diffusers. An average value of ψ is 0.2, although higher values can be realized. This value, however, will be assumed in these calculations.

The dynamic pressure = $\psi \frac{W\bar{U}^2}{2g}$ lb/sq ft

$$\begin{aligned} \therefore p &= \frac{W (\bar{U}^2 - u^2)}{2g} + \psi \frac{W\bar{U}^2}{2g} \\ &= \frac{W}{2g} \left[(1 + \psi) \bar{U}^2 - u^2 \right] \text{ lb/sq ft.} \end{aligned}$$

As the pressure is given, this equation can be used, after substituting for the peripheral velocities to obtain the impeller radius

$$R = \sqrt{\frac{1}{1 + \psi} \left(\frac{1.63p}{n^2 W} + r^2 \right)} \text{ ft} \quad (4)$$

Substituting for r from (1) and inserting the value of $\psi = 0.2$, then

$$R = \sqrt{1.36X + 0.261 Y^2} \text{ ft} \quad (5)$$

where

$$X = \frac{p}{n^2 W} \text{ and } Y = \frac{Q}{vW}.$$

Fig. 7 shows the dependence of R^2 on X and Y .

Pump casings for throughputs below 10 lb/sec, when a maximum of three separate delivery diffusers are used, can be considered as formed of two flat plates of uniform thickness. The pump shown in Fig. 4 could be considered as an example of such a construction.

$$\text{Weight of casing } w_p = k_1 R^2 \text{ lb}$$

where k_1 is a constant.

The weight of the oxidant pump casing with two diffusers shown in Fig. 1 was 2 lb; the impeller has a radius of 0.1 ft, and is made of aluminium alloy as is usually the case. For this pump which may be considered as typical

$$k_1 = 200$$

hence

$$w_p = 200 R^2 \text{ lb} \quad (6)$$

Fig. 8 shows this relationship for throughputs up to 10 lb/sec.

For pumps where more than three diffusers are used it is more convenient to employ a collecting ring forming part of the pump casing into which the fuel or oxidant flows from the diffusers. The total flow is then taken from this ring. The pump shown in Fig. 2 is an example of this type of construction. The mass of the pump is then concentrated at its circumference, hence only the weight of this portion will be considered. For the purposes of weight estimation it is assumed that the thickness of the wall of this collecting ring does not vary much.

The weight of the casing (neglecting the side walls of the pump) is then given by

$$w_p = k_2 R b \text{ lb}$$

where k_2 is a constant and b the impeller width at the blade root. A reasonable figure for b is

$$b = \frac{r}{2} = 0.28 \sqrt{\frac{Q}{vW}} \text{ ft}$$

Substituting this value of b , and the weights and dimensions for the pump shown in Fig. 2, then

$$\begin{aligned} w_p &= 480 R \sqrt{\frac{Q}{vW}} \text{ lb} \\ &= 480 R Y \text{ lb} \end{aligned} \quad (7)$$

Fig. 9 shows the dependence of w_p on R for different values of Y .

2.3 Weight of Pump Impellers

The impeller weights are very small compared with pump weights, but although they do not affect the total weight of the pump materially their weight and positions on the shaft must be taken into account when considering the critical rotational speed of the turbine.

The pump impeller weight is given by:

$$w_I = k_4 R^2 b \text{ lb}$$

where k_4 is a constant.

The weight of the aluminium alloy oxidant pump impeller of the turbo-pump in Fig.1 is 0.046 lb, and therefore, by substitution we obtain

$$\begin{aligned} w_I &= 48.4 R^2 \sqrt{\frac{Q}{vW}} \text{ lb} \\ &= 48.4 R^2 Y \text{ lb} \end{aligned} \quad (8)$$

Fig.10 shows the dependence of the weight of the aluminium alloy impeller on R^2 for different values of Y . If the impeller is of stainless steel the weight obtained from Fig.10 is somewhat greater than this, but it will certainly be less than the ratio of the densities of stainless steel and aluminium alloy (2.8) multiplied by the weight of the aluminium impeller.

2.4 Weight of turbine rotor and shaft

Now consider a simple impulse turbine with one pressure stage when the width of the rotor is more or less constant up to fairly high powers.

An average value for the maximum tip speed is 600 ft/sec.

∴ The radius of the turbine rotor

$$\begin{aligned} R_T &= \frac{600}{2\pi n} \text{ ft} \\ &= \frac{95.5}{n} \text{ ft} \end{aligned} \quad (9)$$

If it is assumed that most turbine rotors will be of stainless steel, then the weight of the rotor alone = $k_5 R_T^2 \rho_s$ lb, where k_5 is a constant and ρ_s is the density of stainless steel. Let the length of each bearing be twice the diameter D of the shaft and suppose also that the overhang to accommodate each impeller and fluid seal is $2D$. Then the weight of the effective length of the shaft = $2\pi D^3 \rho_s$ lb, and the total weight of the turbine rotor and shaft $w_R = k_5 R_T^2 \rho_s + 2\pi D^3 \rho_s$ lb.

The criterion for the diameter of a shaft transmitting power is normally the torque acting on the shaft, but as turbo-pumps run at comparatively high speeds when the torque is low, it is felt that the peripheral speed of the shaft in the bearings is a better criterion. A reasonable figure is 60 ft/sec and in unlubricated carbon bearings this has proved satisfactory. The critical speed of the shaft with the arrangement of bearings considered is usually very high.

$$\begin{aligned} \therefore D &= \frac{60}{\pi n} \text{ ft} \\ &= \frac{19.1}{n} \text{ ft} \end{aligned} \quad (10)$$

Consider the stainless steel turbine rotor and shaft of the turbo-pump (shown in Fig.1), whose weight is 1.72 lb, the density of stainless steel being 486 lb/cu ft then

$$w_R = \left[\frac{0.25}{n^2} + \frac{21}{n^3} \right] \times 10^6 \text{ lb} \quad (11)$$

This relationship is shown in Fig.12.

2.5 Weight of Turbine casing and bearings

By using a similar method to that used for estimating the weight of the turbine rotor and shaft, the weight of the turbine casing is given as $k_6 R_T^2 \rho_c$ lb, where k_6 is a constant and ρ_c is the density of the casing material (Fig.11).

The bearings together with their casings can be considered as two hollow cylinders each $2D$ long, $2D$ external diameter and D internal diameter. The density of the bearings is assumed to be the same as that of the casings hence the weight of the bearings and casing is equal to $3 \pi D^3 \rho_c$ lb. \therefore The total weight of the bearings and casings

$$w_B = k_6 R_T^2 \rho_c + 3 \pi D^3 \rho_c \text{ lb.}$$

Comparing with the turbo-pump (see in Fig.1) whose turbine and bearing casing are of aluminium alloy and weigh 6.4 lb and assuming that in most cases the material used for turbine casings is aluminium alloy, then

$$w_B = \left[\frac{1.1}{n^2} + \frac{10.6}{n^3} \right] \times 10^6 \text{ lb} \quad (12)$$

Fig.12 shows this relationship together with the total turbine weight w_T which is the summation of equations (11) and (12)

$$w_T = w_B + w_R = \left[\frac{1.35}{n^2} + \frac{31.6}{n^3} \right] \times 10^6 \text{ lb} \quad (13)$$

It is apparent that a conventional turbine would be somewhat lighter than the type indicated here as the nozzles are then separate and a high pressure steam manifold round the casing is not required. The bucket type of turbine has been considered, however, for the following reasons:-

- (1) It is much simpler to manufacture
- (2) As shown by initial tests its efficiency is quite good
- (3) The estimated weight will err on the pessimistic side, as it is heavier than the normal type.

2.6 Weight of Fluid Seals, Bolts, Studs, etc.

The pump shown in Fig.1 utilizes Barske fluid seals which have a low weight which is only 2% of the total weight of the pump.

The weight of the bolts, studs, etc., is 8% of the total weight. Therefore, when the total weight of the pump is obtained with the exception of these two items, it is reasonable to suppose that if this figure is multiplied by $\frac{10}{9}$ to allow for such extras, a fair estimate of the total weight of the pump will be obtained.

3 Estimation of Dimensions

The main dimensions of a turbo pump required in the assessment of a rocket motor are its overall length, turbine casing diameter and the diameter of the two pump casings.

3.1 Overall Length

As stated before, the width of the turbine wheel will not vary much with increase in size and if it is assumed that the bearings are close to the turbine wheel as in Fig.1, the total length becomes:

$$L = \ell_T + 2\ell_B + \ell_{p_O} + \ell_{p_F} + 2\ell_S \text{ ft}$$

where ℓ_T is the width of the turbine wheel, ℓ_B is the axial length of each bearing, ℓ_{p_O} is the axial length of the oxidant pump, ℓ_{p_F} is the axial length of the fuel pump and ℓ_S is the axial length of each fluid seal.

ℓ_T can be taken as a nominal figure of 1 in (0.083 ft). If it is assumed that each pump has a right angle bend at its entrance then the length of the bend is a function of the inlet radius r , and the axial length of each pump is only a function of the blade root width b which in turn is a function of r , or

$$\ell_p = k_7 \sqrt{\frac{Q}{vW}} \text{ ft} \quad \text{where } k_7 \text{ is a constant.}$$

ℓ_p for the oxidant pump (Fig.1) is 0.25 ft and $k_7 = 2.6$

$$\therefore \ell_p = 2.6 \sqrt{\frac{Q}{vW}} \text{ ft}$$

If D ft is allowed for the fluid seals, i.e. $\ell_S = D$ ft, but from equation (10)

$$D = \frac{19.1}{n} \text{ ft}$$

then

$$\begin{aligned} L &= 0.083 + \frac{114.6}{n} + 2.6 \left[\sqrt{\frac{Q_O}{v_O W_O}} + \sqrt{\frac{Q_F}{v_F W_F}} \right] \text{ ft} \\ &= 0.083 + \frac{114.6}{n} + 2.6 (Y_O + Y_F) \text{ ft} \end{aligned} \quad (14)$$

3.2 Pump and turbine diameters

The diameter of a pump D_p is proportional to its impeller radius R . The oxidant pump diameter (Fig.1) is 0.354 ft and the impeller radius is 0.1 ft.

$$\therefore D_p = 3.54 R \text{ ft} \quad (15)$$

The diameter of the turbine casing D_T is proportional to the turbine rotor radius R_T , but from equation (9)

$$R_T = \frac{95.5}{n} \text{ ft},$$

and the diameter of the turbine casing of the turbo-pump (Fig.1) is 0.584 ft

$$\therefore D_T = \frac{268}{n} \text{ ft} \quad (16)$$

This does not allow for any projecting inlets or outlets required for the fuel, oxidant or steam which may entail some increase in overall diameters.

3.3 Installation of a turbo-pump in a projectile or fuselage

When considering the variation of dimensions of a turbo-pump with thrust, the three main variables concerned, length, diameter and thrust must be correlated. Consider the cross-section through a projectile or fuselage as shown in Fig.13. The position of the pump as shown is most convenient with regard to inlet and outlet connections, and as can be seen the diameter of the projectile can be taken as the criterion. Fig.14 shows the dependence of projectile diameter on thrust when considering the space required to accommodate comfortably a turbo-pump.

4 Discussion

The methods of weight and size estimation have been based on the similarity of design with existing turbo-pumps and also involve certain simplifying assumptions. Consequently it cannot be expected that they will give weight and dimensions nearer than a first approximation. This is, of course, what is required in determining whether a motor should be designed with pressurised propellant tanks or with a turbo-pump.

The work involved has been reduced to a few mechanical operations and can be summarised as follows:-

- (a) Obtain the rotational speed from equation (3),

$$n = 48.2 \sqrt{\frac{vW}{Q}} \text{ revolutions per second or by using Fig. 3 or 4.}$$

- (b) Determine radii of pump impellers from equation (5)

$$R = \sqrt{1.36X + 0.261 Y^2} \text{ ft, where } X = \frac{p}{n^2 W} \text{ and } Y = \sqrt{\frac{Q}{vW}}$$

or by using Fig.7.

- (c) Determine weight of pump casing from equation (6)

$$w_p = 200 R^2 \text{ lb (for flows below 10 lb/sec) or equation (7)}$$

$$w_p = 480 RY \text{ lb (for flows over 10 lb/sec) or by using Fig. 8 or 9.}$$

- (d) Determine impeller weights from equation (8),

$$w_I = 48.4 R^2 Y \text{ lb or by using Fig.10.}$$

- (e) Determine turbine weight from equation (13),

$$w_T = \left(\frac{1.35}{n^2} + \frac{31.6}{n^3} \right) 10^6 \text{ lb}$$

or by using Fig.12.

(f) Obtain the total and multiply by $\frac{10}{9}$ to allow for fluid seals, studs, bolts, etc.

The dimensions are obtained from the following:-

(g) Total length $L = 0.083 + \frac{114.6}{n} + 2.6 (Y_O + Y_F)$ ft .

(h) Pump diameter $D_p = 3.54 R$ ft .

(i) Turbine casing diameter $D_T = \frac{268}{n}$ ft .

It will be noted that the overall weight for a given propellant mass flow mainly depends upon the rotational speed which is fixed by conditions in the pump with the largest throughput normally the oxidant pump. The higher the oxidant density and the less the oxidant throughput, the higher is the rotational speed and hence the smaller are the dimensions and weight. This can be illustrated by considering turbo-pumps required for the same rocket motor performance, but using different oxidants entailing different fuel/oxidant mixture ratios. Fig. 15 shows the relationship between the weight of the turbo-pump and the motor thrust for the combinations HTP/kerosene and Nitric acid/kerosene. In both cases a specific impulse of 200 seconds has been assumed and mixture ratios of 11/1 and 5/1 respectively, but because of the superior oxidant density characteristic and the small oxidant throughput of nitric acid/kerosene the pump unit is 18% lighter.

A type of turbo-pump whose rotational speed is much higher than that indicated by calculation from paragraph 2.1, is described in Appendix II. The weight of such a pump is consequently considerably reduced.

5 An estimation of turbo-pump steam consumption

It is rather difficult to obtain an accurate estimate of the steam consumption of a turbo-pump, but it is desirable to have an idea at least of what it might be in order that the designer may obtain an estimate of the amount of extra propellant (e.g. hydrogen peroxide) he will require to drive the turbo-pump.

Consider the overall efficiency of a turbo-pump.

$$\eta_o = \frac{\text{H.P. Output of pumps}}{\text{H.P. Input of steam to turbine}}$$

The H.P. output of each pump = $\frac{Q_p}{550W}$.

The H.P. available in the steam is given by

$$E_T = \frac{J \times \text{Heat drop available} \times \text{flow rate of steam in lb/sec}}{550}$$

$$= 2.54 H_r q_s \text{ where } H_r \text{ is the heat drop in CHU/lb and } q_s \text{ is steam flow rate lb/sec.}$$

The heat drop can be obtained from the entropy diagram for hydrogen peroxide steam. The steam pressure ratio and exhaust temperature will affect the heat drop, but as an approximation, it is 150 CHU/lb over a range of pressure ratio 20 to 25 at an exhaust temperature of 375°K.

$$\therefore \eta_o = \frac{\left(\frac{Q_o p_o}{W_o} + \frac{Q_{FPF}}{W_F} \right)}{2.54 \times 150 \times q_s \times 550}$$

or

$$= \frac{4.75}{q_s} \left[\frac{Q_o p_o}{W_o} + \frac{Q_{FPF}}{W_F} \right] \times 10^{-6} \quad (17)$$

Representative figures for existing turbo-pumps are given in the following Table I.

TABLE I

Steam Consumption of Turbo-pump

Turbo-pump	Type of Turbine	Number of Stages	Steam	Temp. °C	Steam Cons. (lb/sec)	Overall Eff. %
Beta I	Bucket	One press One velocity	HTP	450	0.625	8
109-509	Normal	One pressure (steam reversed back through blades)	HTP	450	1.0	12.5
A4(V.2)	Normal	One pressure Two Velocity	HTP	450	Approx. 5.8	15.5

As far as increasing the efficiency is concerned the Beta I turbo-pump has not been fully developed yet, but it is felt that 10% can be achieved.

If this figure of 10% overall efficiency is assumed when considering the steam consumption of a turbo-pump in a proposed fuel supply system for a rocket motor, a good approximation will be obtained as follows:-

$$q_s = 47.5 \left[\frac{Q_o p_o}{W_o} + \frac{Q_{FPF}}{W_F} \right] \times 10^{-6} \text{ lb/sec} \quad (18)$$

Fig.16 shows the variation of steam consumption with rocket motor thrust, for the two fuel/oxidant combinations considered in paragraph 4.

6 Conclusions

This note indicates methods by which the designer of a rocket motor can obtain the weight, dimensions and steam consumption of a possible turbo-pump for that motor. The estimates are rather conservative and are based on present knowledge, but it is pointed out that with improvements in both turbine and pump design the calculated figures can be reduced. Appendix II shows a method of reducing turbo-pump weight. The use of idling shroud pumps² will also make a great improvement in overall efficiency of the turbo-pump. If, therefore, this note indicates that there is not a great difference between a certain pressurized system and a turbo-pump system for the same duty, the designer is recommended to investigate the turbo-pump design in more detail.

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2	A.D. Baxter	High Pressure Pumps for Rocket Propellants. Technical Note No. R.P.D.12.

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APPENDIX I

List of Symbols

v	-	velocity of propellant at pump inlet - ft/sec
p	-	pump delivery pressure - lb/sq ft
Q	-	pump delivery - lb/sec
q_s	-	steam flow rate - lb/sec
W	-	propellant density - lb/cu ft
\bar{U}	-	peripheral velocity at outer radius of impeller - ft/sec
u	-	peripheral velocity at inner radius of impeller - ft/sec
n	-	rotational speed - revolutions per sec
R	-	pump impeller radius - ft
D_p	-	pump overall diameter - ft
r	-	pump inlet radius - ft
b	-	width of impeller blade at root - ft
R_T	-	turbine rotor radius - ft
D_T	-	turbine overall diameter - ft
D	-	diameter of shaft - ft
D_F	-	diameter of projectile or fuselage - ft
L	-	total length of turbo-pump - ft
ℓ_T	-	width of turbine wheel - ft
ℓ_B	-	axial length of bearings - ft
ℓ_p	-	axial length of pump - ft
ℓ_s	-	axial length of fluid seals - ft
w_p	-	weight of pump - lb
w_I	-	weight of pump impeller - lb
w_R	-	weight of turbine rotor and shaft - lb
w_B	-	weight of turbine and bearing casing - lb
w_T	-	total weight of turbine - lb
ρ_s	-	turbine rotor and shaft material density - lb/cu ft
ρ_c	-	turbine casing material density - lb/cu ft
η_o	-	overall efficiency of turbo-pump
E_I	-	horse power available in steam

APPENDIX II

Means for Reduction of Turbo-pump Weight

If the rotational speed of a turbo-pump can be increased and other factors such as pressure, flow rates, etc., remain the same, the overall diameter can be reduced. In order to keep the impeller blade length at a reasonable figure the inlet diameters also have to be decreased, but the reduction in size is limited by the dependence on the velocity of flow through the inlet, which should not exceed 10 ft/sec as indicated in paragraph 2.1. This is applicable to the oxidant pump since it always has the greatest delivery. If the inlet were divided into two branches either two impellers in one pump are used or one wider impeller with an entry both sides is used, then much smaller impeller diameters and consequently higher speeds could be utilized. A turbo-pump with the former arrangement of oxidant pump is shown in Fig.17. This pump would be slightly heavier and run at the same speed as the pump shown in Fig.1, but it would deliver twice the quantity of propellants.

To obtain the rotational speed of such a pump the quantity Q in equation (3) can be halved, so that the speed would be $\sqrt{2}$ times that of the normal type of pump with the same delivery.

The various weights can then be calculated as indicated before, with the exception that the weight of the oxidant pump should be doubled, to allow for the extra complications. The length will not be much affected.

Difficulties may be encountered in designing a small diameter turbine for high powers, but it should be possible to use either two turbine wheels on one shaft, or if using a bucket turbine utilizing the double bucket as shown in Fig.15.

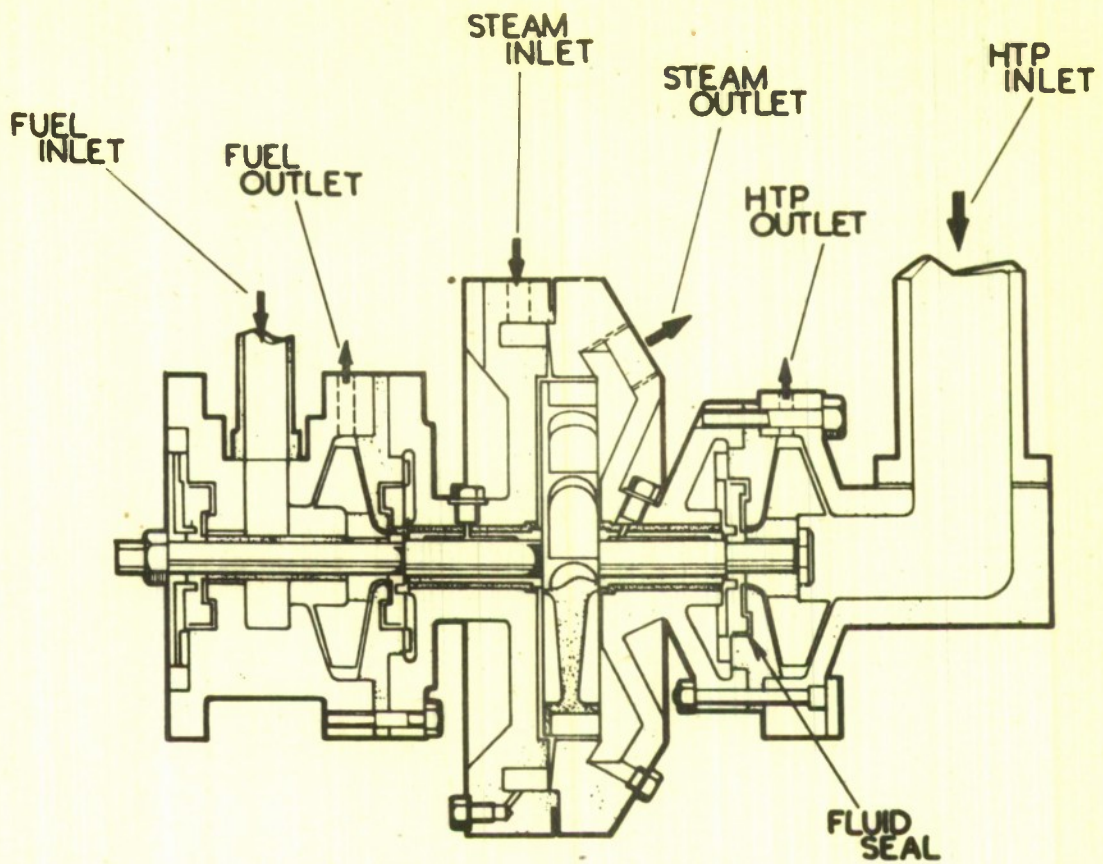


FIG. 1. DIAGRAM OF TURBO PUMP

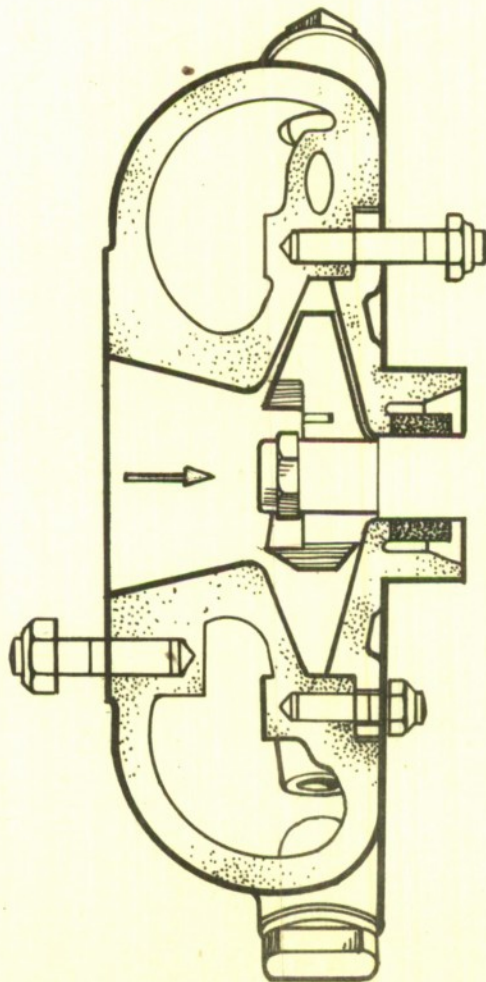


FIG. 2. HIGH FLOW RATE OPEN IMPELLER PUMP

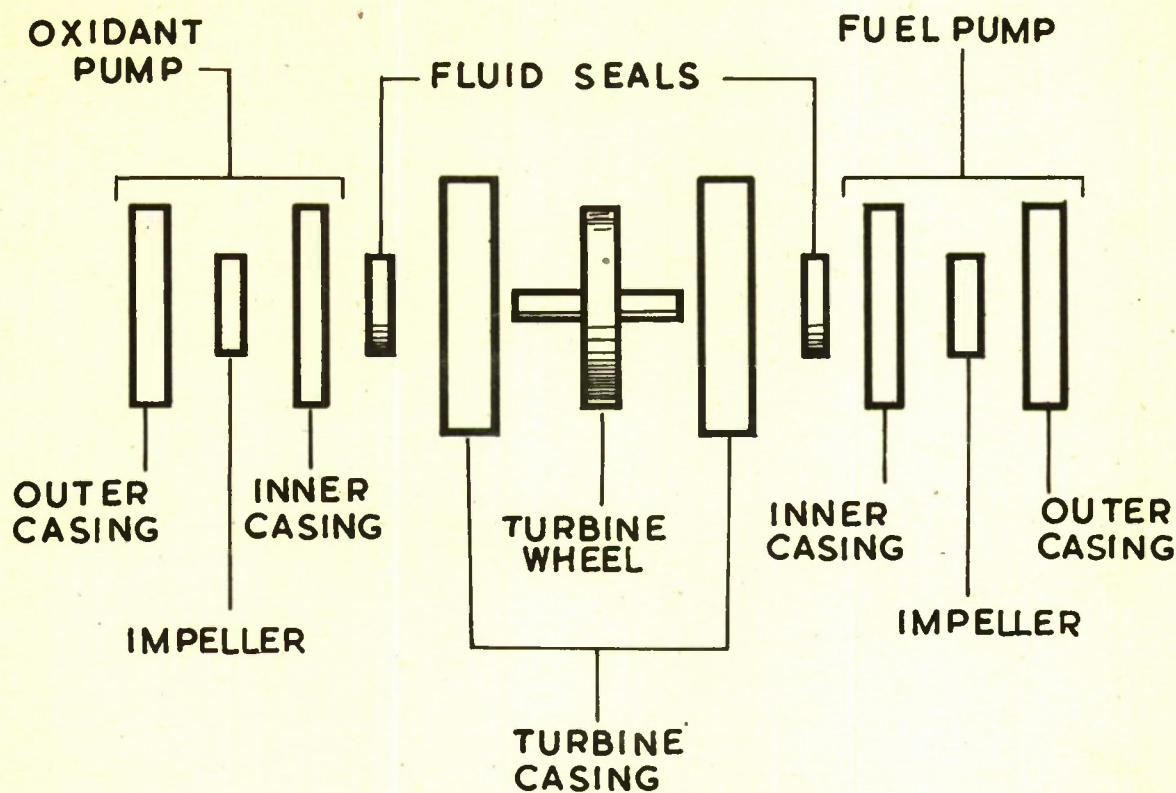


FIG.3. TURBO PUMP. EXPLODED VIEW SHOWING COMPONENTS

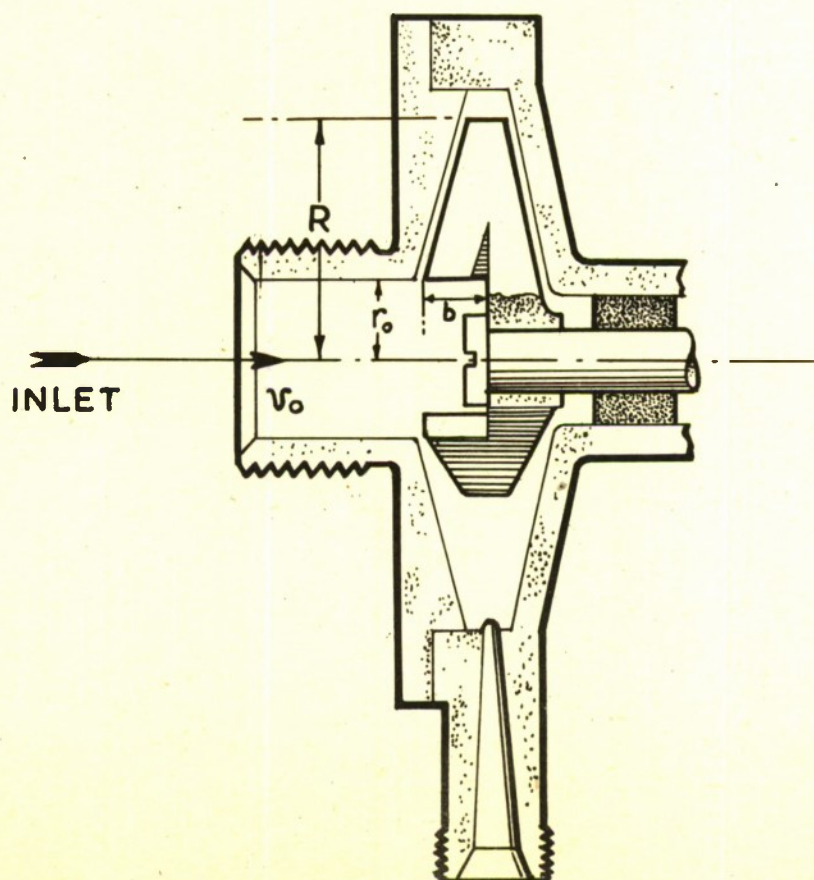


FIG.4. DIAGRAM OF OPEN IMPELLER PUMP

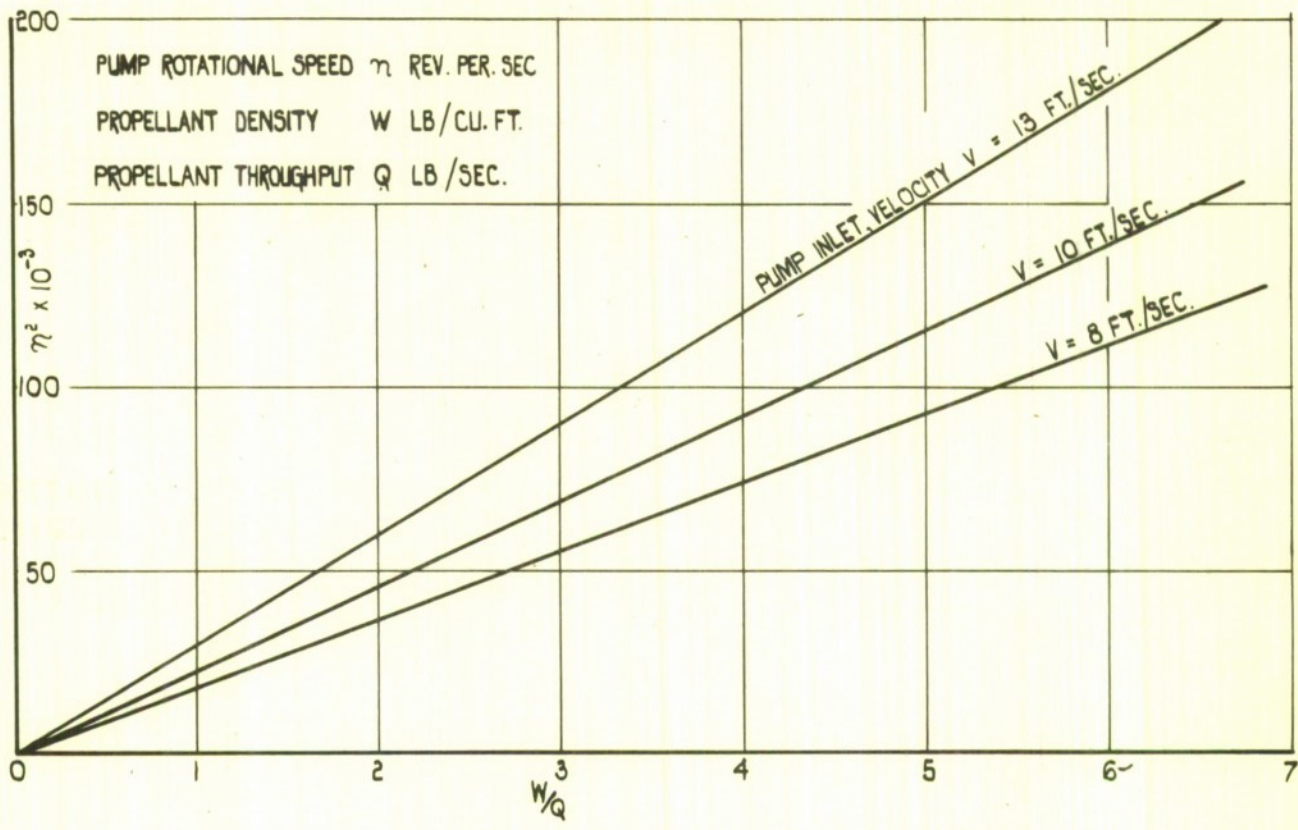


FIG.5 ESTIMATION OF TURBO-PUMP ROTATIONAL SPEED
(PUMPS OF LOW THROUGHPUT)

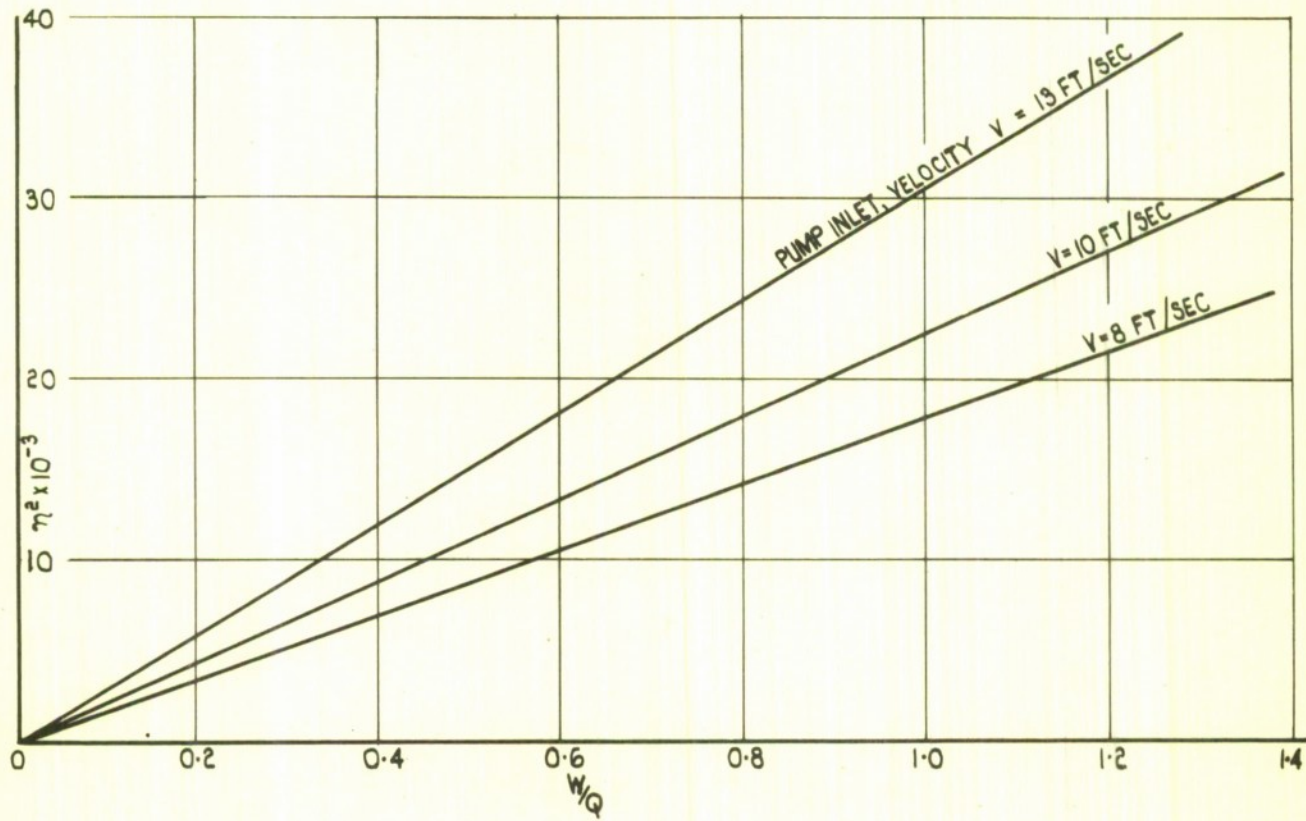


FIG.6 ESTIMATION OF TURBO PUMP ROTATIONAL SPEED
(PUMPS OF HIGH THROUGHPUT)

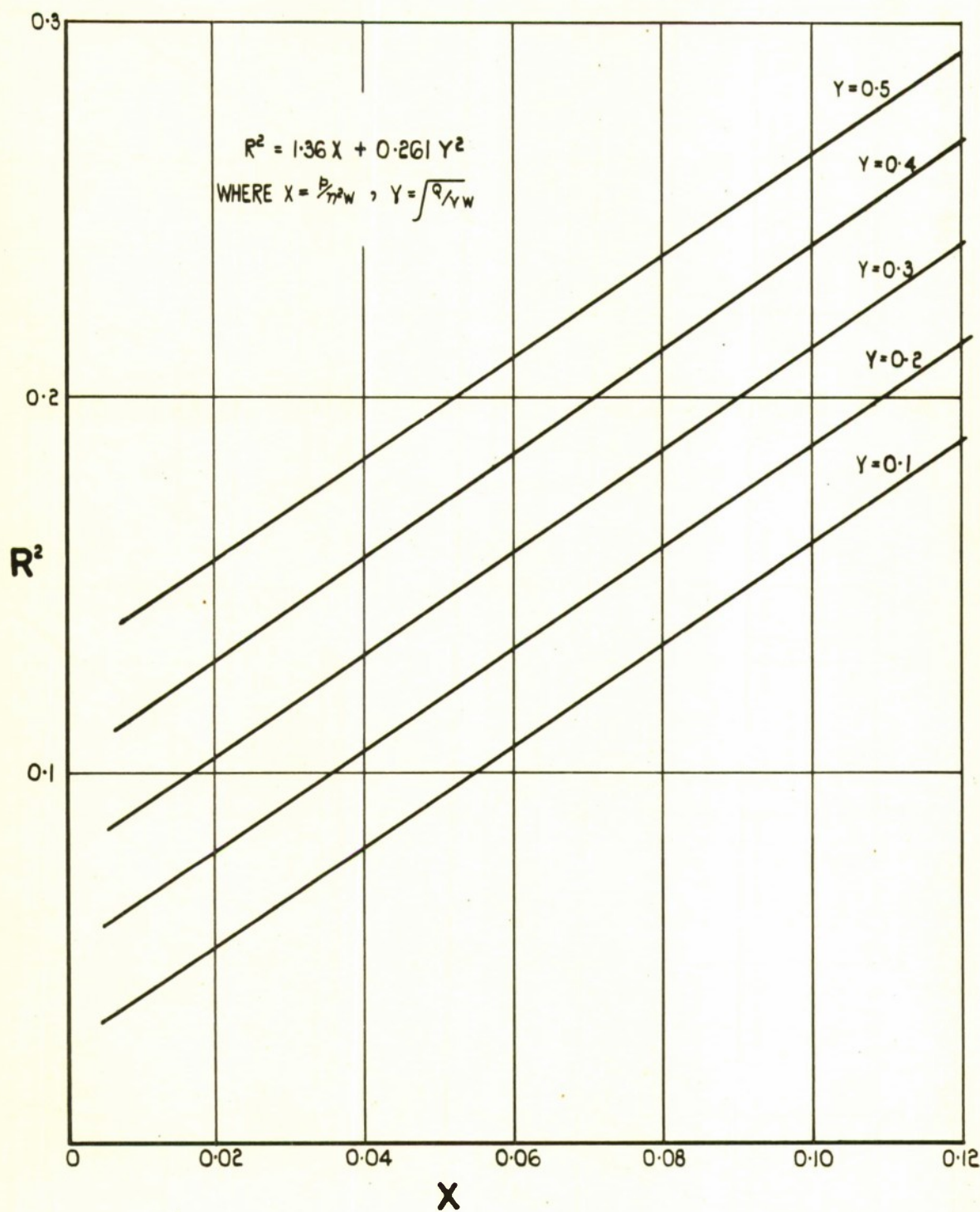


FIG. 7 DEPENDENCE OF IMPELLER RADIUS ON X AND Y

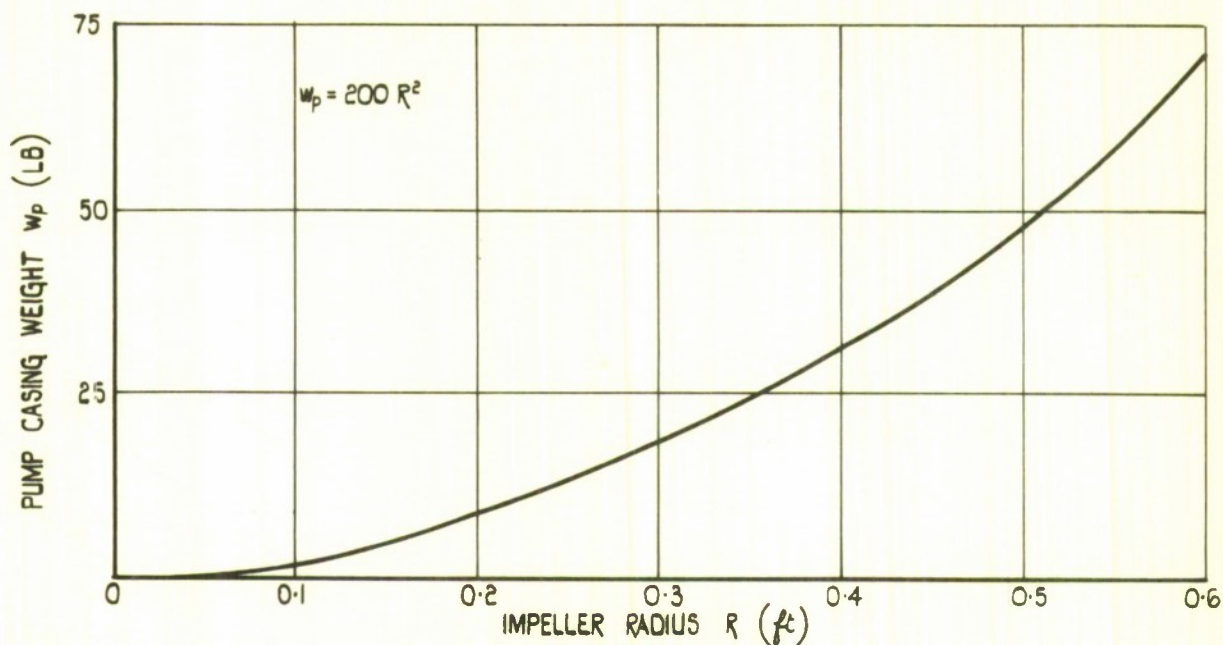
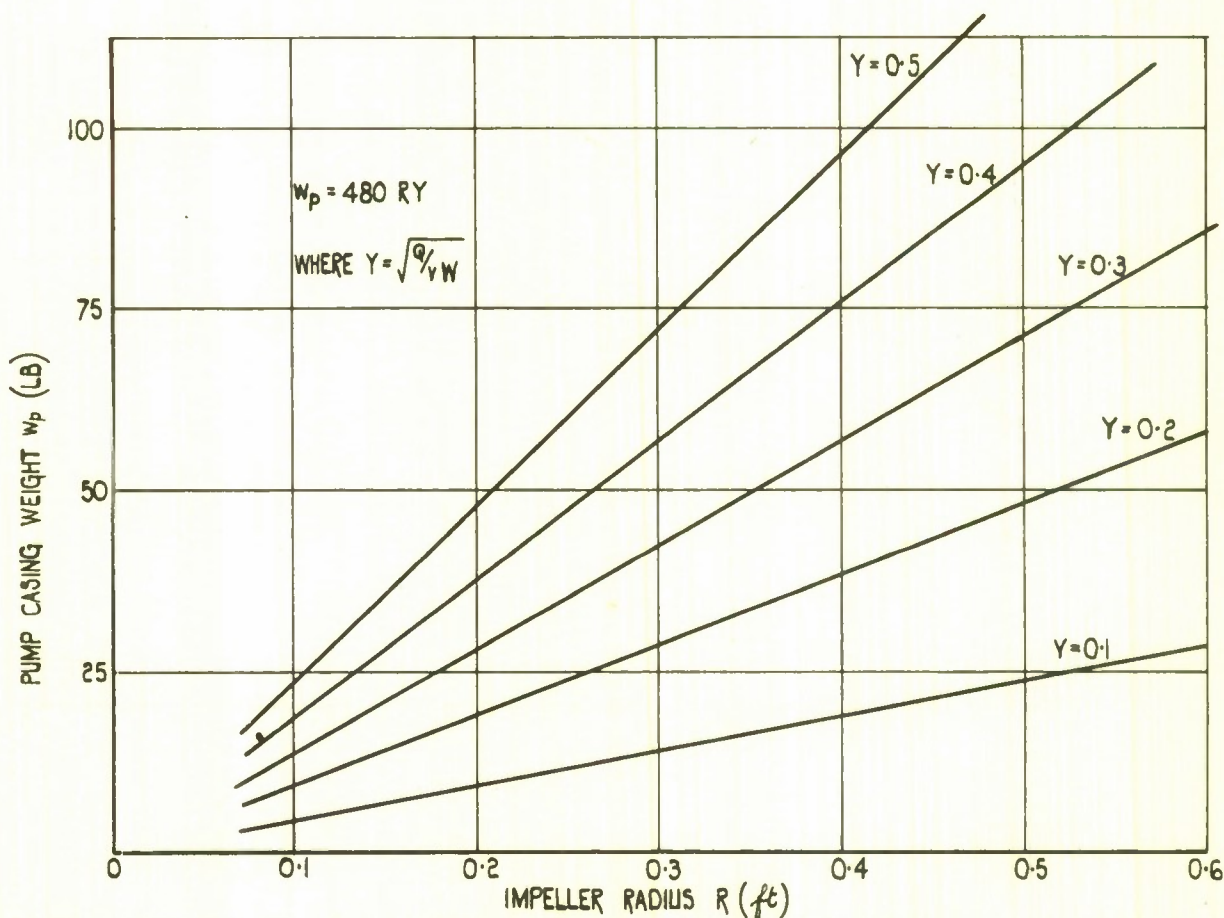


FIG. 8 DEPENDENCE OF PUMP CASING WEIGHT ON IMPELLER RADIUS.
(FOR THROUGHPUTS BELOW 10 LB/SEC.)



FIGS. 9 DEPENDENCE OF PUMP CASING WEIGHT ON IMPELLER RADIUS.
(FOR THROUGHPUTS ABOVE 10 LB/SEC.)

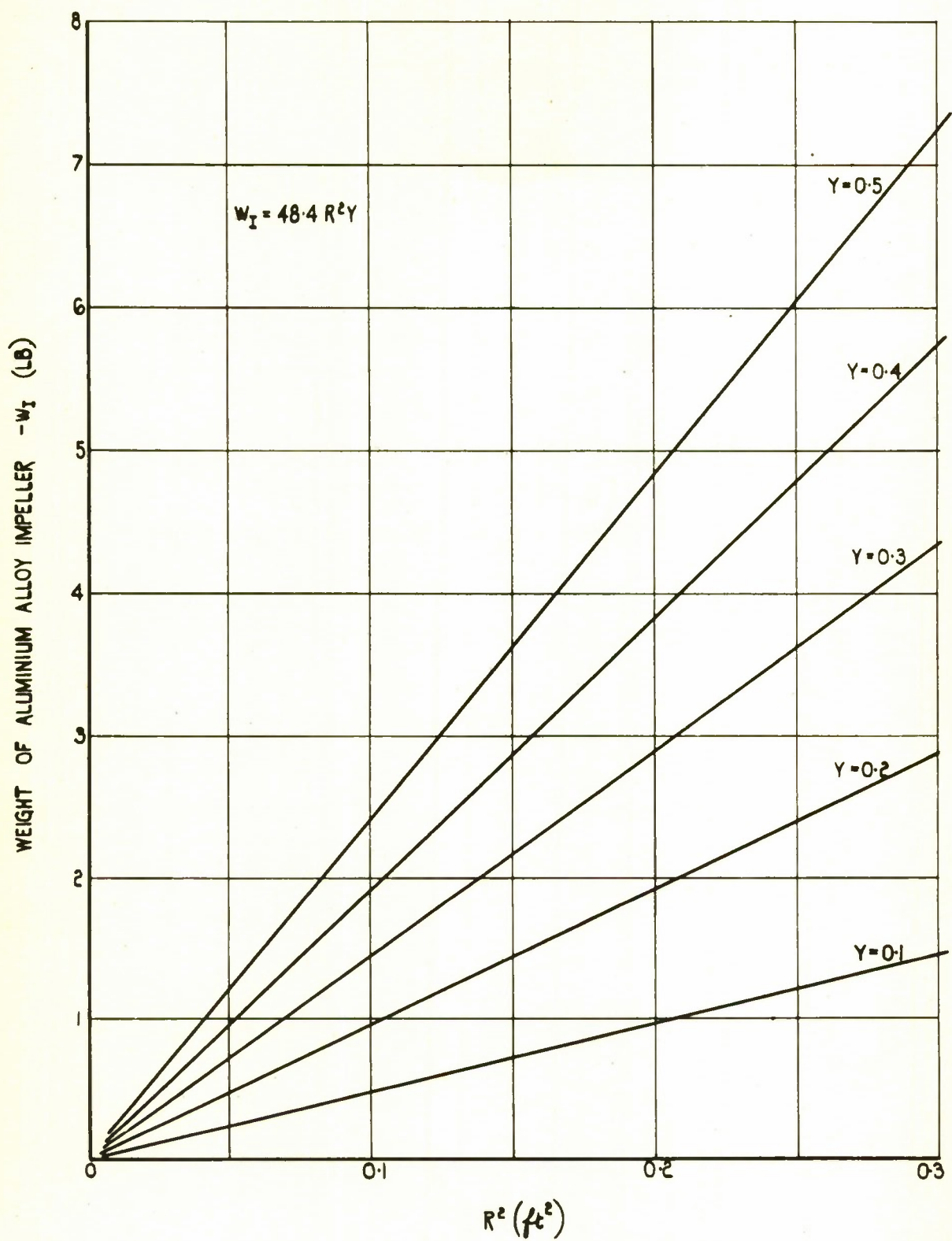


FIG. 10 DEPENDENCE OF ALUMINIUM ALLOY IMPELLER
WEIGHT ON IMPELLER RADIUS.

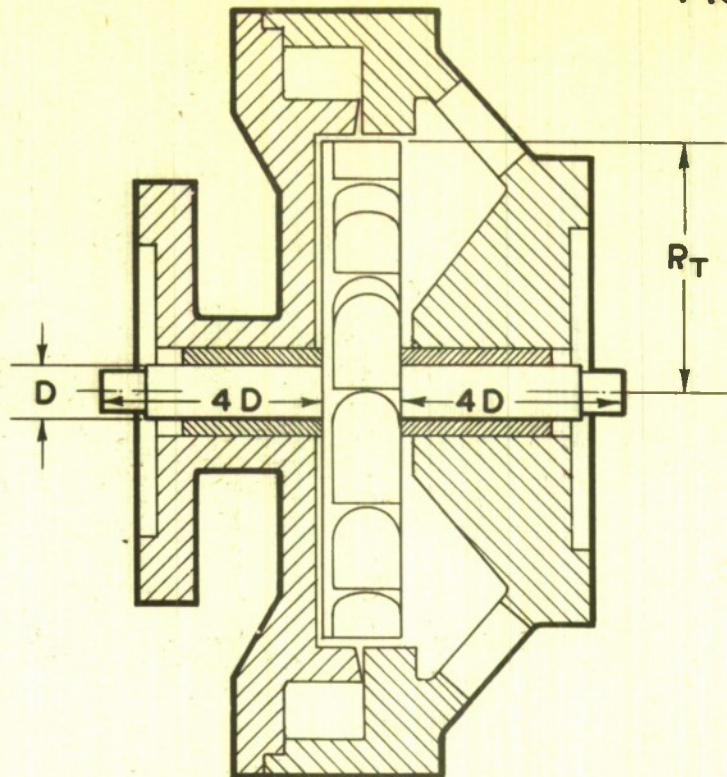


FIG. 11. TYPICAL TURBINE AND CASING

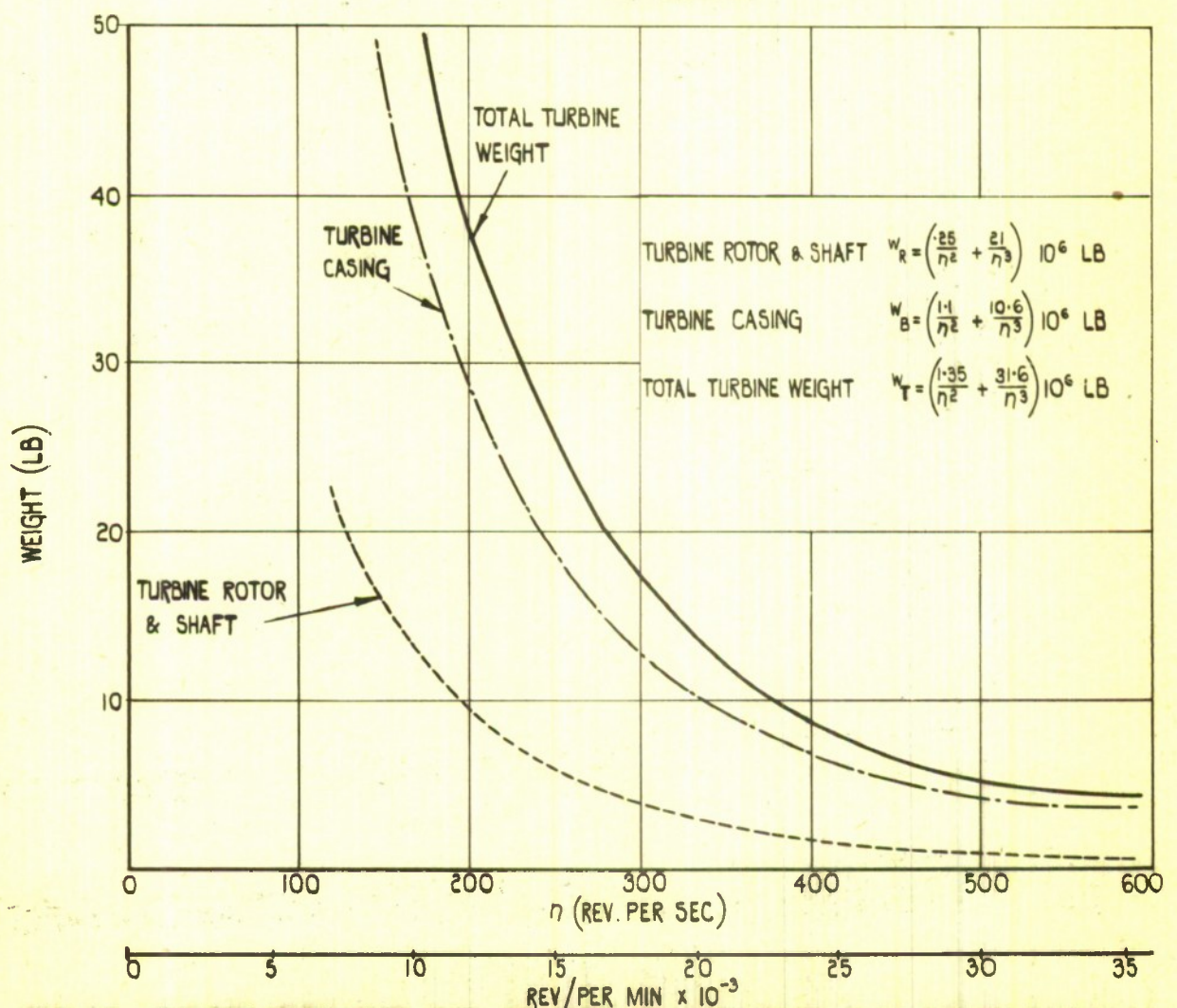
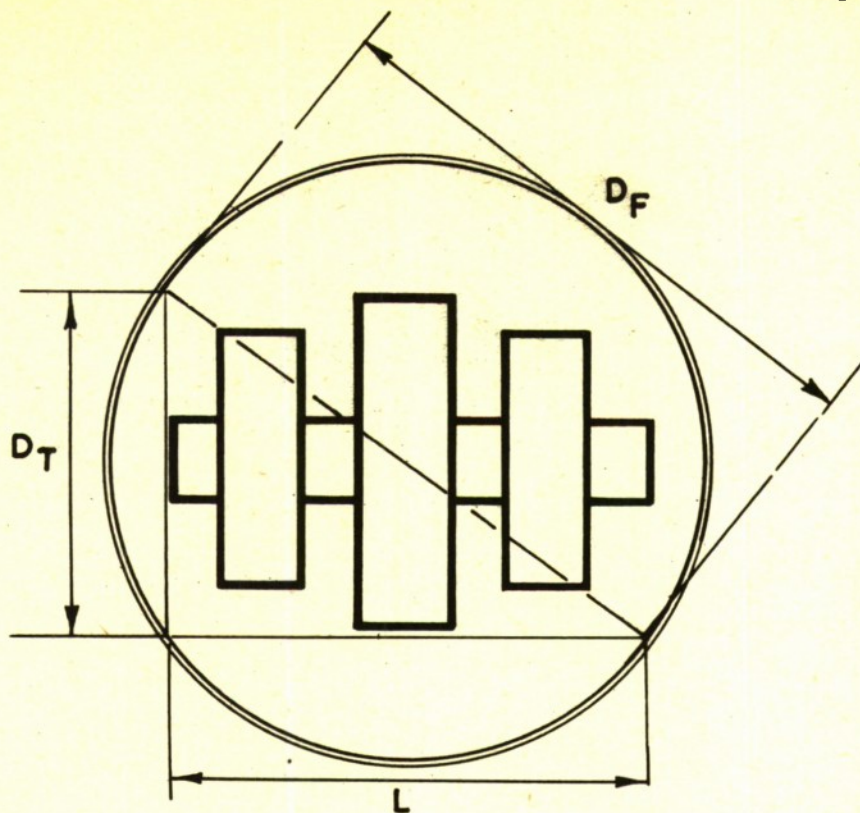


FIG. 12 DEPENDENCE OF TURBINE WEIGHT ON ROTATIONAL SPEED



$$D_F = \sqrt{D_T^2 + L^2}$$

FIG. 13. INSTALLATION OF A TURBO PUMP IN A PROJECTILE OR FUSELAGE

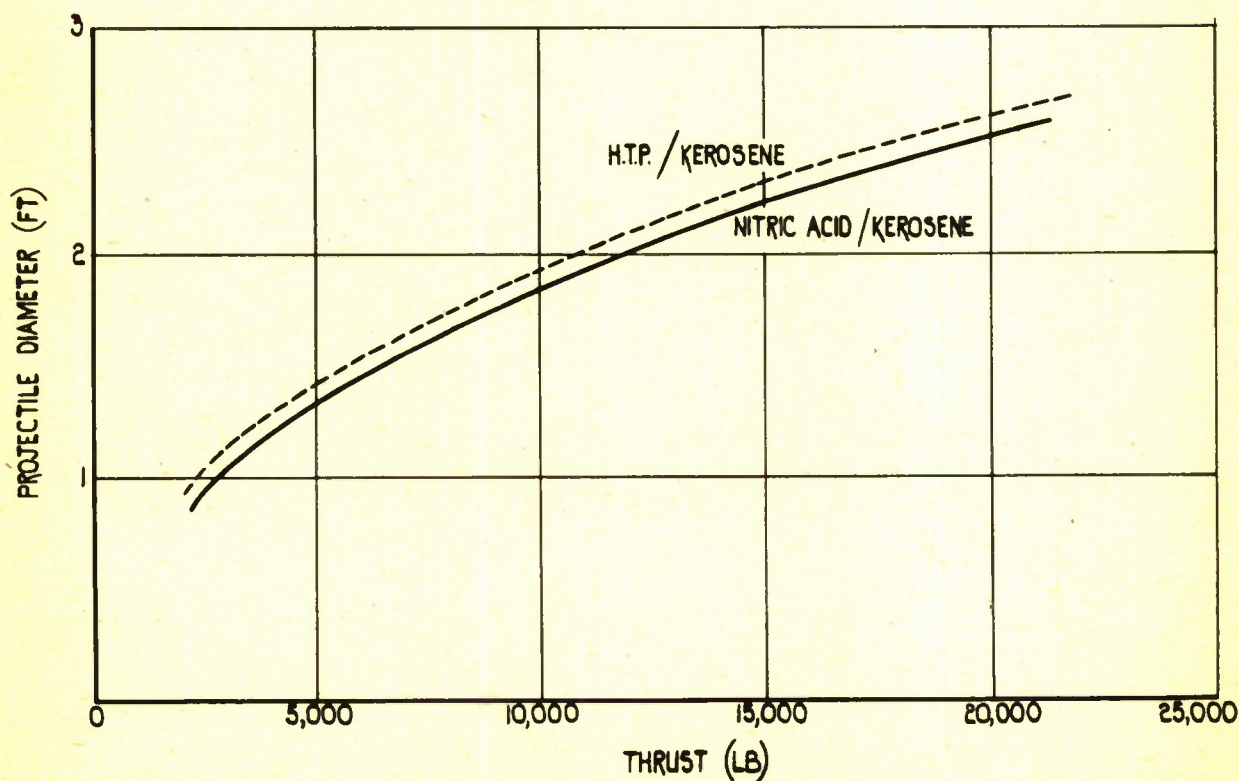


FIG. 14 DEPENDENCE OF MINIMUM PROJECTILE DIAMETER ON THRUST.

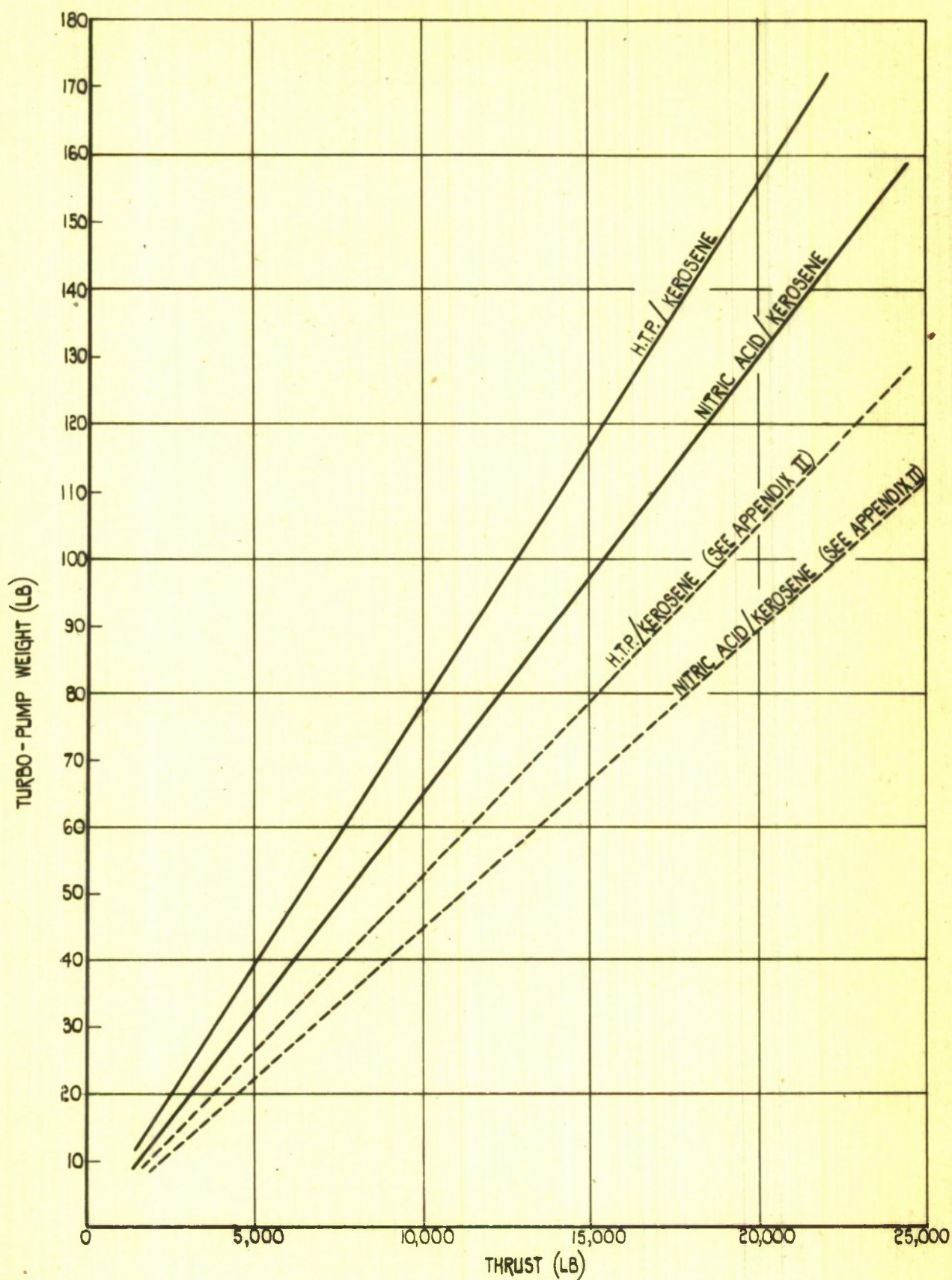


FIG.15 DEPENDENCE OF TURBO PUMP WEIGHT ON THRUST.

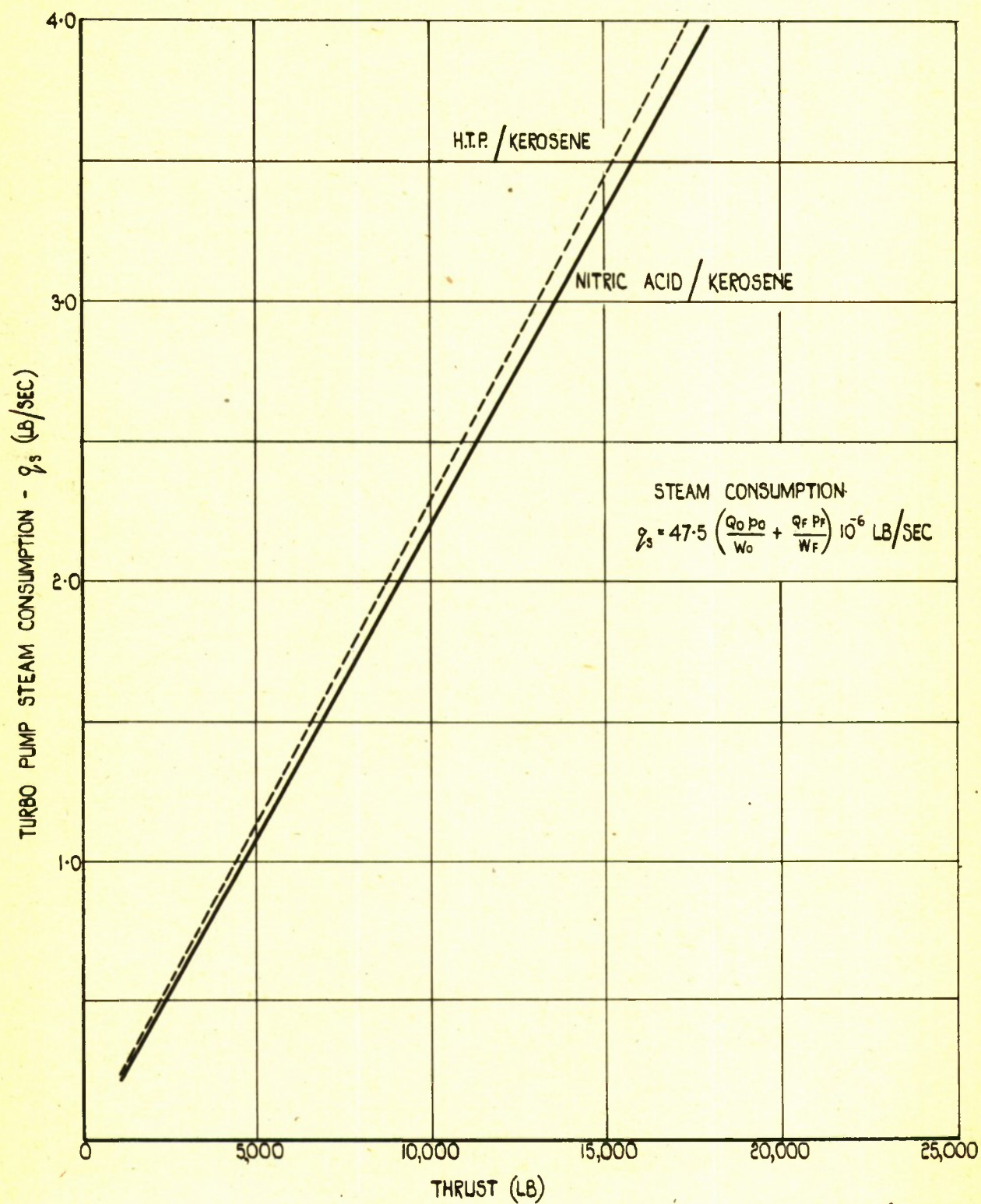


FIG.16 DEPENDENCE OF TURBO PUMP STEAM CONSUMPTION ON THRUST.

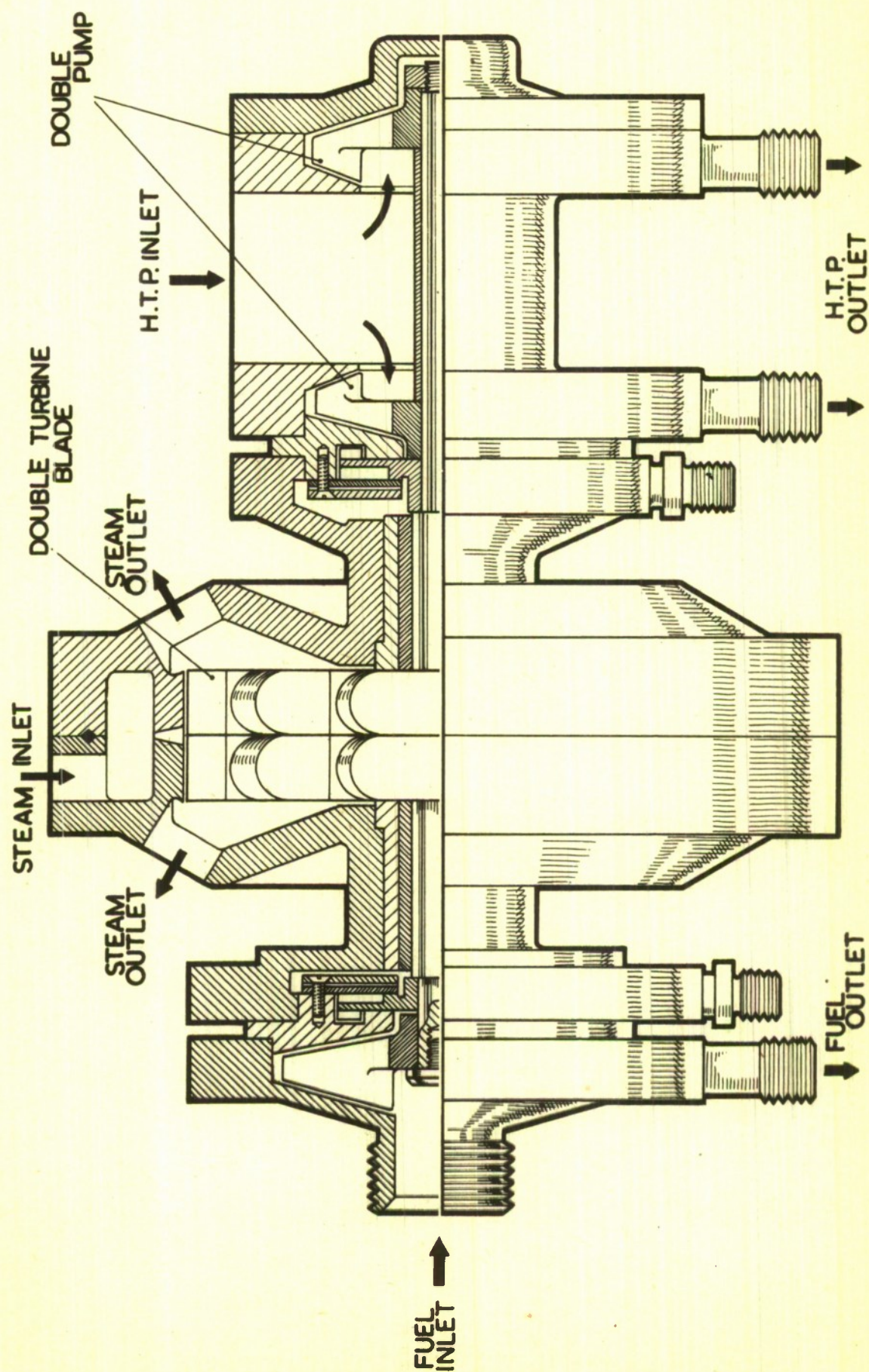


FIG.17. TURBO-PUMP FOR HIGHER THROUGHPUT



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